AD 402 738

6333

Monthly Progress Report for September 1961

on the Continuation of the 3-KW Stirling

Cycle Solar Power Systems Program

Contract No. AF33(616)-8332

Project No. 3145, Task No. 30500

Project Engineer: D. S. Monson
Phone: CHapel 4-1511, Extension 4866

Engineering Department Report No. 2323

H. D. Wilsted

Approved: T. F Nagey
for Director of Research

03

Allison Division
General Motors Corporation
Indianapolis, Indiana

ALLISON

TABLE OF CONTENTS

INTRODUCTION

This report summarizes all work performed under Contract No. AF33(616)-8332 during the month of September 1961.

ALLISON

I. PROBLEM ELIMINATION TEST PROGRAM

MODEL PD-46 ENGINE

No engine tests were run during the month of September. Work in this period was directed toward the evaluation and interpretation of the data previously obtained and procuring modified components to complete the series of performance tests.

ENGINE TEAR-DOWN INSPECTION

This engine inspection followed 30 hours, 12 minutes of engine running, during which the mechanical operation of the engine was satisfactory. At the time of shut-down there was no specific mechanical trouble apparent. Inspection revealed several minor areas which require further investigation and testing. With the exception of the following variations, the engine was in completely satisfactory condition.

Piston Shaft Seals

Slight oil leakage was found to have occurred through both the displacer piston shaft seal and the power piston shaft seal. The leakage through the power piston shaft seal amounted to approximately 20 cc of oil, while that through the displacer shaft seal was an indeterminate amount that had adhered to the piston and gas passages. This was partially caused by the reuse of the damaged displacer shaft seal from the previous build-up because a new seal was not available.

Slight scoring of the shafts in the area of the guide bushings was also noted. Indications are that these shafts will require a higher surface hardness for extended operation. All future shafts will be hardened by carburizing the seal and bushing area.

Regenerators

The regenerators were removed from the cylinder head for complete inspection. Each cartridge weight was compared to that before testing and it was found that the results were inconclusive in that most showed weight gains of approximately 0.05 grams while two indicated a weight loss of 0.06 grams each. Since these two weights were obtained on different scales (one at Harrison Radiator Division and the second at the Allison Division) it was not possible to recheck the initial weight.

Visual inspection revealed a deposit on the screen mesh on the cold side of each regenerator. See Figure 1. This deposit is believed to be a combination of oil and wear residue from the Teflon K-30 piston rings. A flow test at teardown indicated that the pressure drop through the side arm circuit increased approximately seven percent when compared with the check made at Build-up 7. This deposit was removed by ultrasonic cleaning followed by conventional cleaning processes.

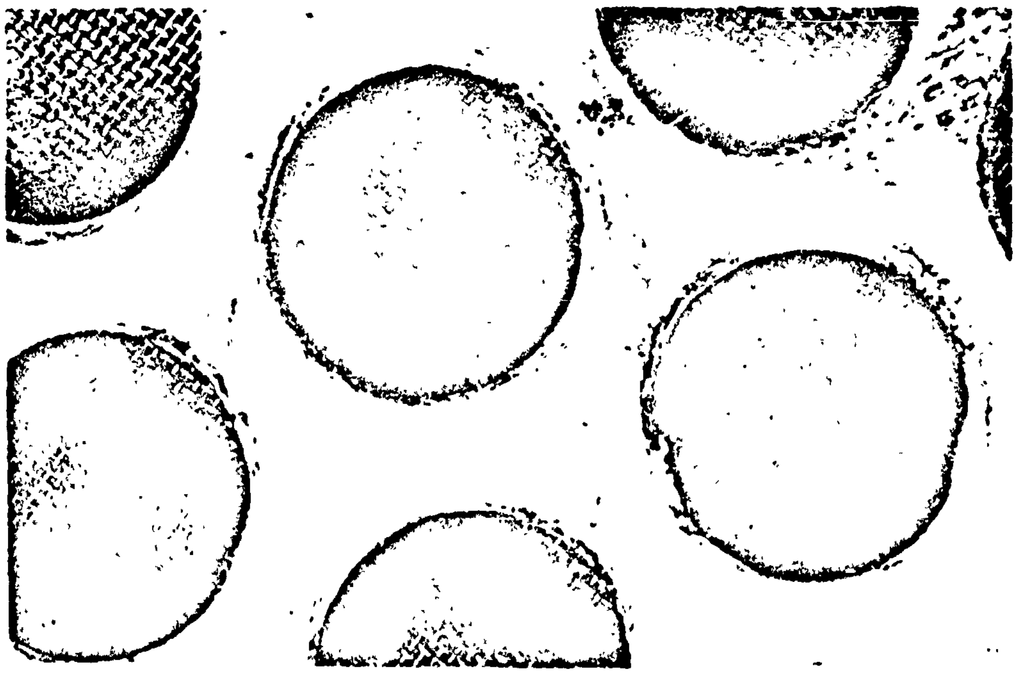


Figure 1. Deposits on Screen Mesh on Cold Side of Regenerator

Figure 2. Screen Mesh Damage on Hot Side of Regenerator

Damage to the screen was observed on the hot side of the regenerator where heater tubes coincided with regenerator end plate holes. See Figure 2. This is felt to have been caused by bombardment of small foreign particles in the working fluid against the screen which had been weakened by the 1100°F temperature.

A solution to this problem is more thorough cleanliness during assembly.

Generators

Slight generator rub was present, but was not severe enough to cause trouble. The high spots on the stator have been stoned down to prevent further rubbing. Two pieces of tape, used to hold wires during fabrication, had been loosened by oil. See Figure 3. Delco-Remy Division has assured Allison that it was an oversight that the tape had not been removed previously and that so long as the wires were rigid the tape is not required. All such tape has now been removed from both generators.

MODIFICATION AND ENGINE BUILD-UP NO. 8

The transfer piston has been reworked to provide a thermocouple clearance well and a reduced clearance white metal seal band. This was required to provide a more accurate measurement of the expansion zone temperature and reduce losses caused by blow-by of the transfer piston.

The water cooler housing was reworked to allow the pressure phase angle transducer to be mounted directly in the gas stream. This should eliminate the lag of the pressure signal encountered during the last test.

Three white metal pistons were fabricated and prepared for testing. One piston was sprayed by the Philips method and two were plated by an Allison process with different percentages of MoS_2. The sprayed piston will be installed and tested initially.

All sharp corners in the gas passages were rounded to reduce flow losses.

Only partial assembly was completed due to unexpected delays in rework of the various components. New shaft seals were installed and the crankcase assembled to the point where it was necessary to await parts from the shop.

FUTURE PLANNING

During October, the assembly of the engine will be completed. The tests to be performed are as follows.

1. Run a one by four performance test (four different pressures at design speed and four different speeds at design pressure) using helium and combinations of helium and argon as the working fluid.

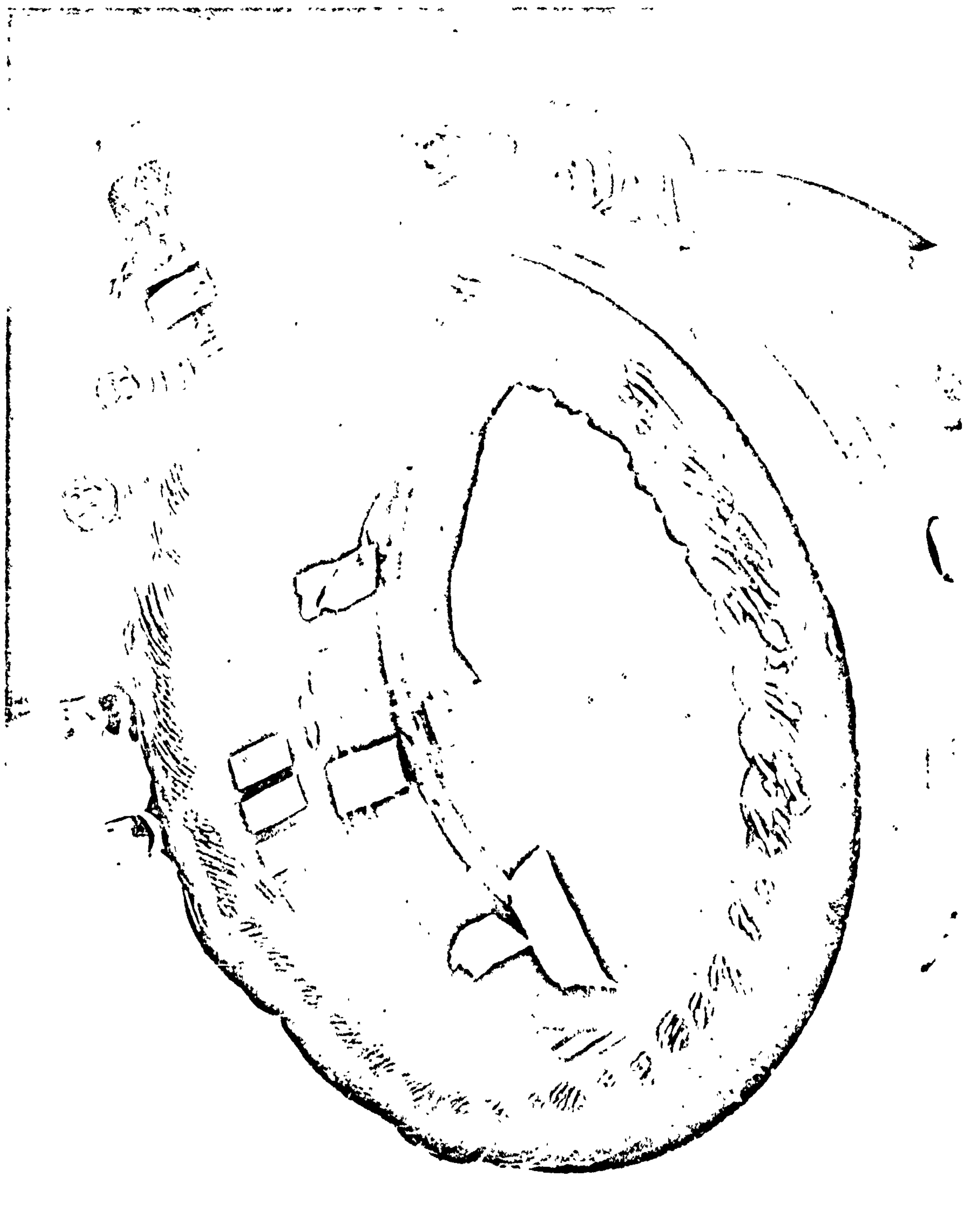


Figure 3. PD-46 Generator

2. Recheck the phase angle with improved transducer location.

3. Measure the hot gas temperature above the transfer piston.

4. Run a test to determine the engine starting requirements.

5. Complete the control test work previously initiated.

6. Run a short test to determine the quality of the electrical output. This information is required to determine flight type engine flywheel requirements.

PD-46 PERFORMANCE

Several theories have been explored for improving the performance. These theories are basically all concerned with improving the heat transfer coefficients in the heater and cooler. Although Allison's past measurement of gas temperature in the expansion space have shown thermocouple readings of 1150°F at the design point, an unfavorable condition existed in the location of the thermocouple which may have affected the accuracy of the reading. The thermocouple was installed flush with the cylinder wall, instead of protruding into the hot space. This permitted the thermocouple to be heated by wall conduction and radiation without much chance of the gas removing this heat effectively. Steps have been taken to correct this condition.

The evidence that leads to the suspicion that the heater temperature is low is that the heat input is considerably below the theoretical value and the heat rejection is slightly above its theoretical value. Figure 4 shows that decreasing heater temperature has exactly this characteristic—that is, the heat input decreases rapidly with decreasing temperature and heat rejection increases slowly. With the placing of the actual values on this curve, it can be seen that a heater temperature of approximately 900°F would give the measured performance.

A low heater gas temperature could be caused by one or more of the following:

1. Low heat transfer coefficient on the NaK side

2. High thermal resistance in the tube wall or any coating in conjunction with it

3. Low heat transfer coefficient on the gas side

The NaK side heat transfer has been tested by varying the NaK flow rate and observing the change in an overall heat transfer coefficient. This test indicated that the heat transfer was not limited by the NaK side. Several uncertainties in the evaluation of this test do not eliminate consideration of this possibility and the design of PD-67 will incorporate features to improve the flow condition of the NaK over the tubes and higher velocities will be used.

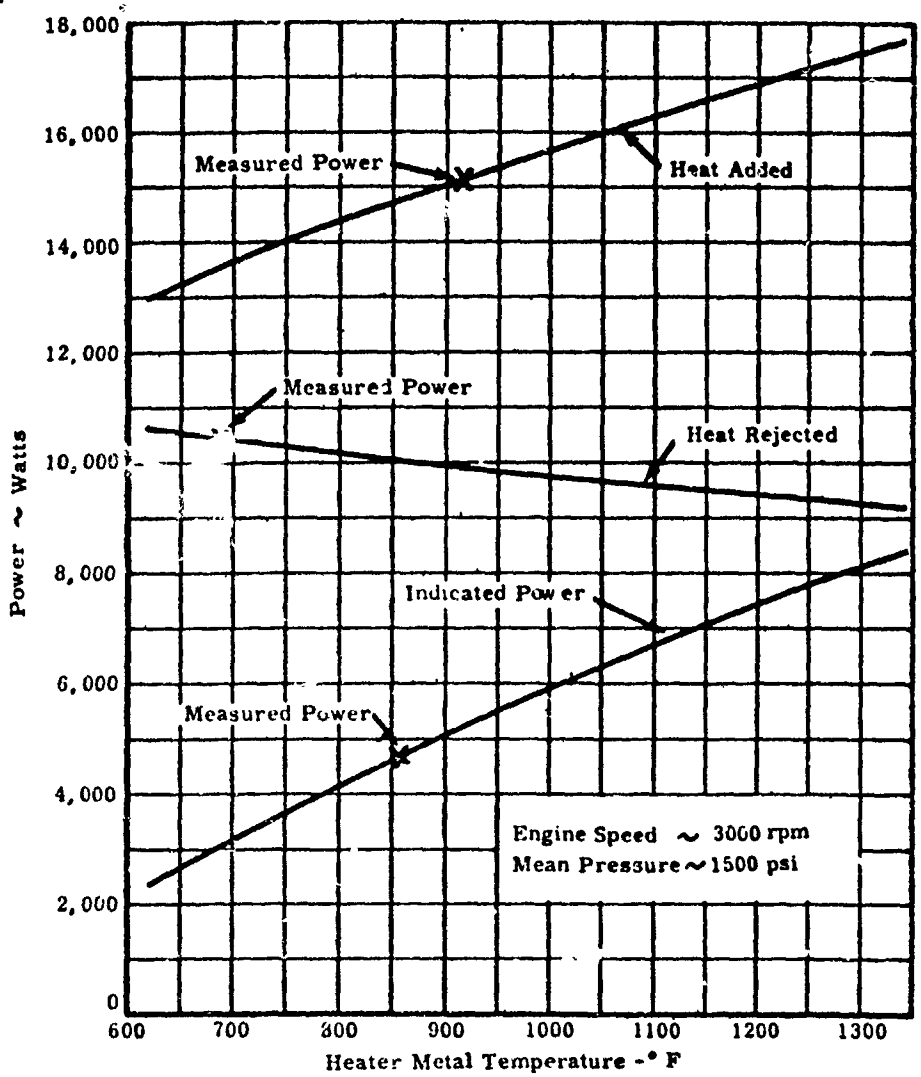


Figure 4. PD-46 Stirling Engine Theoretical Performance

The temperature drop through the heater tube wall has been calculated to be 17°F. A layer of low thermal conductivity material either on the inside or outside of the tubes could increase this temperature drop. Although this possibility is considered to be slight, it should be kept in mind.

The data that was used to estimate the gas side heat transfer coefficient was taken from Reference 1 for the laminar flow regime and Reference 2 for turbulent flow. Since

transition normally occurs between Reynolds numbers of 2000 and 10,000, the data in the transition region was obtained by fairing the data from both sources as shown in Figure 5. The mean Reynolds number and tube l/d ratio for PD-46 is indicated on the curve.

As is apparent from Figure 5, the mean Reynolds number is in the transition region, where the heat transfer coefficient is quite uncertain. In fact, if the most pessimistic value corresponding to the laminar flow is used, this could easily explain the descrepancy between theoretical and actual performance. If one considers the disturbed flow condition at the tube entrances, it would hardly seem likely that laminar flow could exist at Reynolds numbers of 8000. On the other hand, one must remember that a high frequency flow oscillation, with complete stoppage and reversal of flow taking place, could conceivably alter this situation.

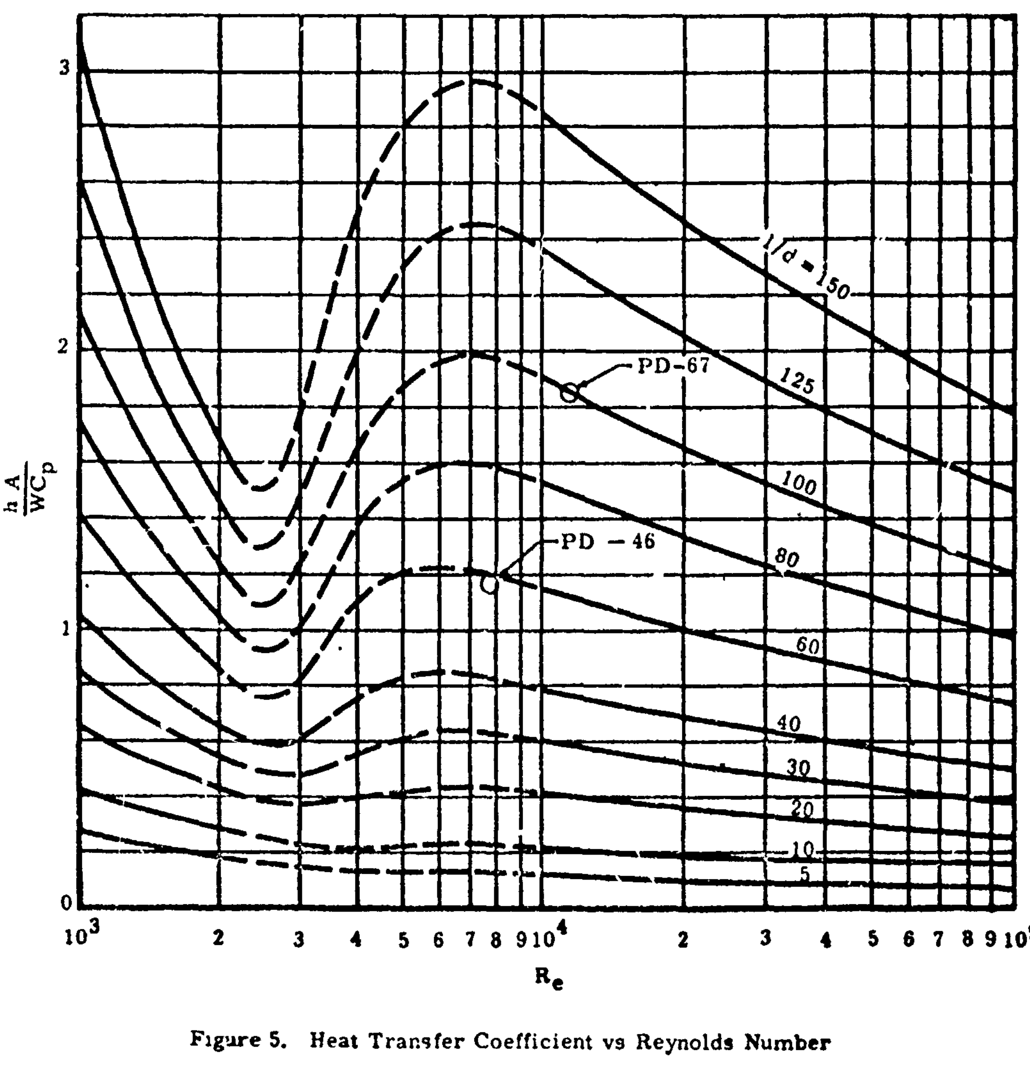


Figure 5. Heat Transfer Coefficient vs Reynolds Number

Many investigators (such as those in Reference 3) have studied pulsating flow, but not with stoppage and reversal. The authors of Reference 3 have been consulted as to their views on this subject and they have agreed that the possibility does exist that transition could be delayed in such a flow situation.

During past tests using nitrogen as a working gas, the mean Reynolds number was 60,000, which is definitely turbulent. Although the flow losses were much higher with nitrogen, when allowance is made for this, it is found that the actual performance is closer to theoretical than with helium. This can be seen in Figure 6, where the power difference is plotted against engine speed. It is felt that improved performance was obtained because

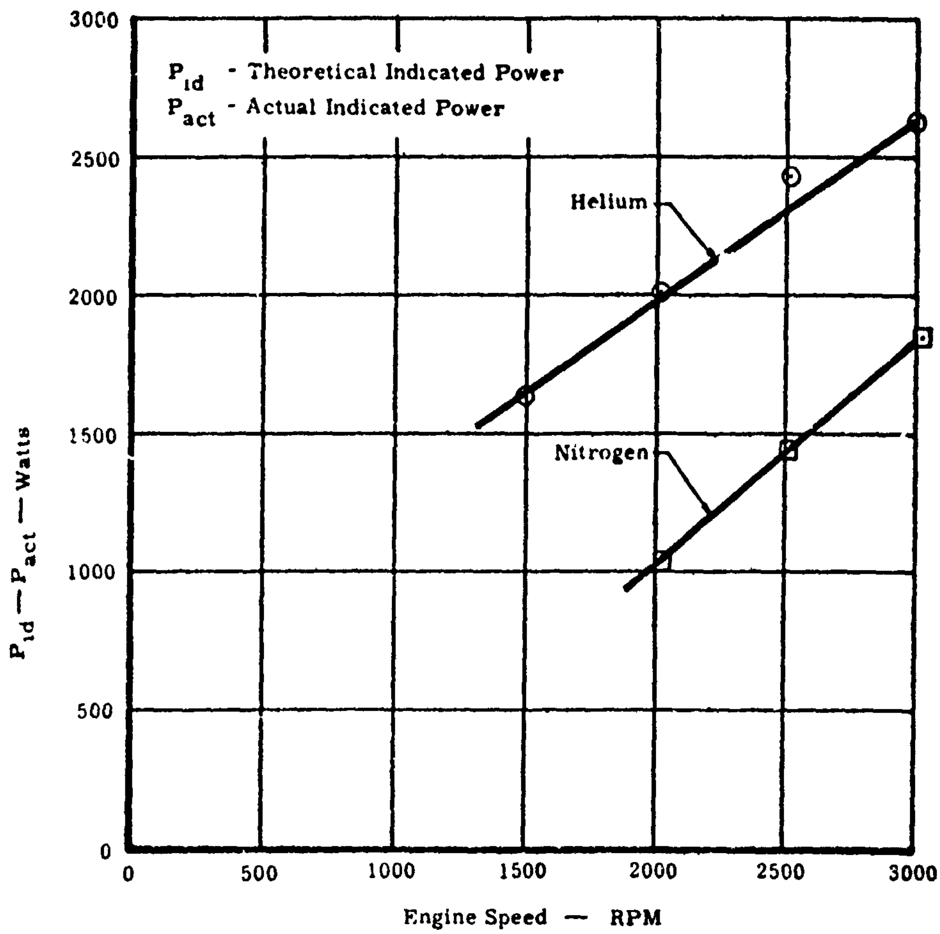


Figure 6. PD-46 Performance with Helium and Nitrogen Working Fluids

the Reynolds number was high enough to ensure turbulent flow. The remainder of the loss with nitrogen remains to be explained.

The nitrogen test was conducted before the improvements in performance measuring techniques were incorporated, so that definite conclusions cannot be drawn from this test.

Future tests are being planned to use mixtures of helium and argon to increase the Reynolds number and to determine if any improvement is made in coming closer to the theoretical performance. Argon was selected as it is monatomic and, therefore, has the same specific heat ratio as helium. It is desirable to maintain the specific heat ratio constant in order to reduce the number of variables.

If delayed transition is causing difficulty, it is relatively simple to increase the Reynolds number in future designs. For the PD-67, a mean Reynolds number of 12,000 and a tube l/d of 100 has been tentatively selected to improve the gas side heat transfer coefficient.

It should also be mentioned that a high gas temperature in the engine cooler could also produce the same characteristics as a low heater gas temperature. This possibility is considered slight, however, since the measured cooler gas temperature is fairly close to the theoretical value. The location of this thermocouple is much more favorable than in the hot end and the reading is considered to be much more realistic. The small amount that the reading is high has been attributed to low water velocity around the tubes, which is being corrected on the PD-67.

Tests of piston seals on the seal test rig have shown that it is possible that ring friction, if present in the engine, could be as high as 2000 watts. The heat from this friction should have shown up as a large increase in engine cooler heat rejection and would not explain the decrease in heat addition. Future tests will be conducted with the white metal piston seal, which should eliminate the seal friction and provide better information on the cycle performance. Attempts will also be made to measure the indicated cycle output with the improved transducer position.

REFERENCES

1. Norris and Streid, Transactions ASME, August 1940, p. 525.
2. McAdams, Heat Transmission, 3rd edition, p. 226.
3. Siegel and Perlmutter, Heat Transfer for Pulsating Laminar Duct Flow, ASME 61 - SA - 28.

II. ENGINE COMPONENT INVESTIGATION

SEAL DEVELOPMENT PROGRAM

Testing of seals throughout the month of September has consisted of screening various seal configurations and materials to determine how these affect leakage and friction. Testing of the PD-46 engine has indicated that the power required for the ring-type piston seal (including friction and power lost through leakage) is of larger magnitude than previously suspected. For this reason, much more attention is being given these parameters prior to initiating endurance testing of a seal. This month, two configuration tests were completed and a third is one-half completed.

Five-Groove Piston with Two-Piece Ring Seal Assemblies

This seal consisted of a steel piston with five 0.090 wide ring grooves. Incorporated into each groove was a two-piece ring assembly. Each ring assembly consisted of one outer square cut ring made of Rulon A material and one back-up spring made of precision L-605 material. The end cut of the outer ring was indexed 180° from the back-up ring cut.

For these tests, the rings had an average room temperature groove side clearance of 0.0005 in. and the back-up springs produced 16 psi loading. These two parameters were selected to produce a minimum of leakage and torque respectively.

This seal was run in a type 440C stainless steel cylinder which was hardened and tempered to Rockwell C55 and had a surface roughness of six micro-inch RMS. The cylinder was round within 0.0001 in. and straight within 0.0002 in.

The cylinder bore diameter was oversize by 0.005 in. and the piston diameter was undersize by 0.010 in. This test was of relatively short duration and for this reason no attempt was made to determine wear rate.

These tests were conducted by installing a single ring assembly into the piston and increasing mean head pressure while operating at constant rpm. At the completion of this test, a static leak check was taken. This procedure was repeated for two, three, four, five, and zero number of rings. The results from these tests are shown in Figure 7.

As can be seen in Figure 7, the static leakage was excessive at all times.* The leakage rate generally decreased with number of rings with the exception of five rings for low pressure differentials. As pointed out earlier, the cylinder was oversize by 0.005 in. This produced an increase in end gap of 0.015 in. The oversize cylinder in combination with a 0.010-in. undersize piston produced a radial clearance which was 0.015 in. larger than necessary. These two dimensions serve to produce a flow area five times that desired and largely account for the excessive static leakage.

---

*K max = 0.3

ALLISON

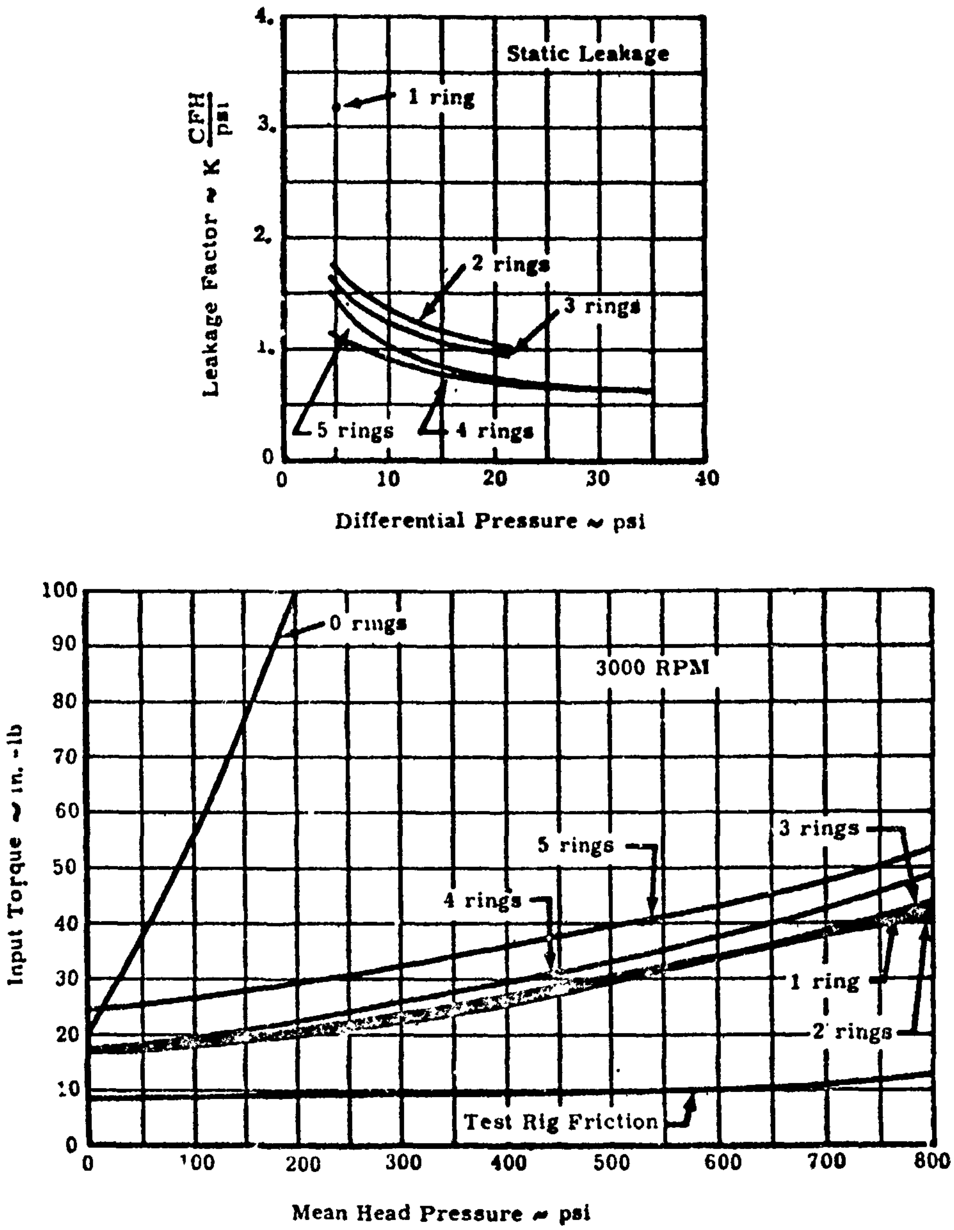


Figure 7. Piston Ring Performance—"Rulon A" Rings with Square End Cuts

The torque characteristics shown in Figure 7 indicate that this seal was producing a very good dynamic seal. The torque curve with zero rings shows the effect of extremely high leakage, which is obviously significant. With the addition of a single ring, the input torque is very near its minimum value. This is due to effecting a considerable improvement in leakage without an appreciable increase in friction. The minimum torque condition at operating pressure occurred with two rings; the addition of rings above this point caused an increase in input torque. This is due to the fact that the improvement in sealing is small compared to the increase in friction. From this investigation it would appear that the seal which would produce the largest power output from the engine should contain two or three rings.

Three-Groove Piston with Two-Piece Ring Seal Assembly

This seal consisted of a steel piston with three 0.180-in. ring grooves. Incorporated into each groove was a two-piece ring assembly. Each ring assembly consisted of a single outer ring made by Koppers of their K-30 material (glass filled Teflon) and had a lap cut. Behind the outer ring was a steel back-up spring. The end cuts between the outer and inner rings were indexed 180° from each other. During operation no direct leakage path exists through the ring.

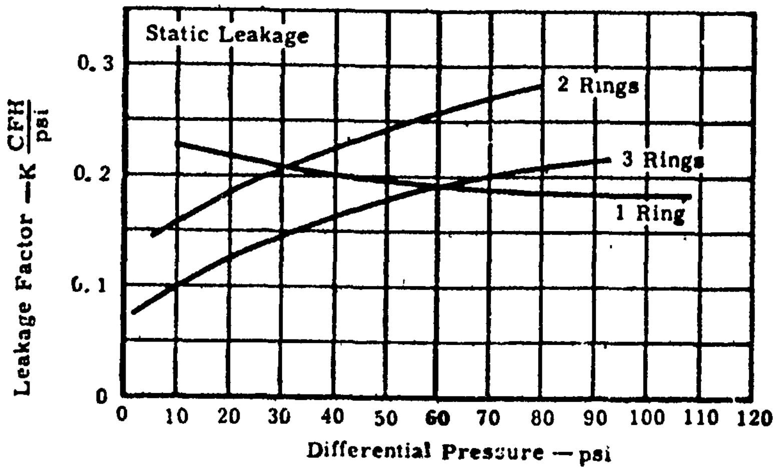
These rings were fitted with an average room temperature clearance of 0.001 in., and a spring producing 16 psi load was used. These parameters were selected to produce a minimum of leakage and torque, respectively. This ring configuration is identical to that used in previous PD-46 engine testing with the exception that the back-up spring for these tests produced one-half of the loading of the engine configuration.

This seal was run in a type 440C stainless steel cylinder which was hardened and tempered to Rockwell C55 and had a surface roughness of three micro-inch RMS. The cylinder was round within 0.0001 in. and straight within 0.0001 in. and was of the correct nominal diameter. This test was of a relatively short duration and for this reason no attempt was made to determine wear rate.

These tests were conducted identically to those for the five-groove piston, and the results are shown in Figure 8.

As can be seen in Figure 8, the static leakage was within the desired maximum value* for all number of rings. For low pressure differentials the leakage rate was as expected, i.e., decreasing leakage with increasing number of rings. Sealing with one ring had a tendency to decrease with increasing pressure and with two or three rings it increased with

*K max = 0.3



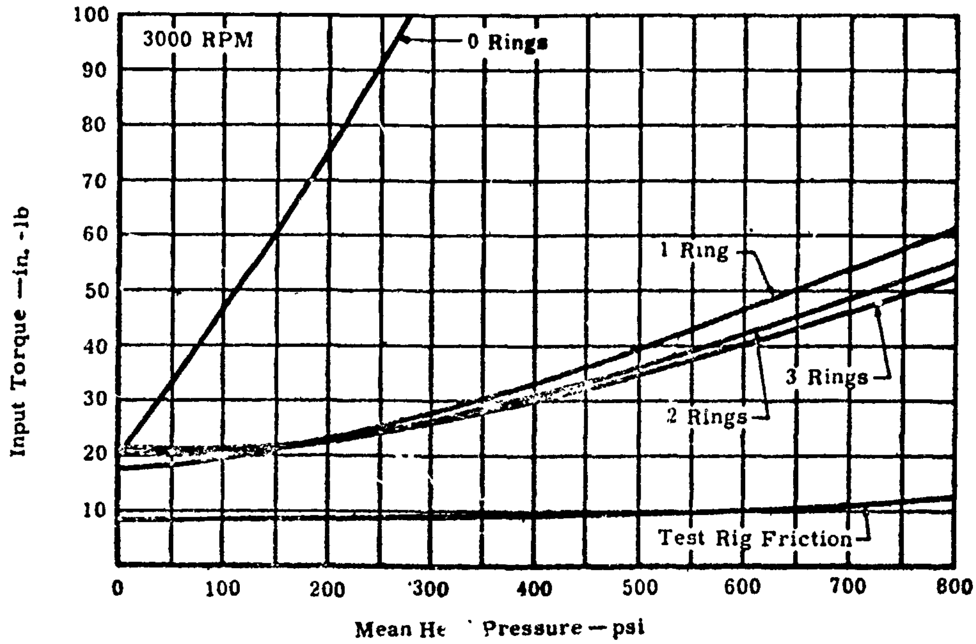


Figure 8. Piston Ring Performance—"Koppers K-30" Rings with Lap End Cuts

increasing pressure.

The torque characteristics were similar in nature to the earlier test; however, the torque was higher at high values of working pressure (i. e., 52-62 in. -lb vs 41-52 in. -lb). The face that this seal produced a higher torque value and that the torque always decreased with increasing number of rings leads one to suspect that this configuration did not produce as fine a dynamic seal as the earlier test configuration although the static leakage was far superior. The effect of smaller piston clearance may be seen for the ringless configurations by comparing Figure 7 to Figure 8.

This seal has a definite tendency for "pumping" at higher pressure; this pumping produced mean pressure differences between the head and buffer zone approaching 200 psi.

Future Plans

The configuration which is now on test is identical to the first test noted in this report except that the cylinder and piston nominal diameters have been brought to the correct values. In addition to obtaining curves as shown in Figure 7, a family of curves with pressure constant and varying speed will be obtained. It is anticipated that the relative magnitude of friction and leakage can be determined this way.

Two additional tests of this nature are planned and will start at the completion of the test now in process. These are:

1. Three-piece ring assembly. This seal is identical to three-piece assemblies used in the past with the exception that the rings have radial step cuts.

2. Four-piece ring assembly. This seal is similar to that noted in Item 1 except three outer step cut rings will be used per groove.

The most promising configuration determined by these screening tests will be placed on endurance testing.

Rings made of Rulon R-S will be available the middle of October. Three end cut configurations are being produced for these tests. (1) 0.090 in. wide ring with a double lap cut, (2) 0.180 in. wide ring with a double lap cut, and (3) 0.090 in. wide ring with a radial step cut. These parts will be given a screening test at the completion of the endurance test noted previously.

UNIT HEAT EXCHANGER TEST

The unit heat exchanger test, which is now completed and analysis of the data in process, was continued with the modified instrumentation as discussed in the last monthly progress

report. Pressure differentials were obtained in the cylinder, at the bottom of the cooler, in the space between the cooler and regenerator, and in the space between the regenerator and the heater tubes for all test runs. Data was taken at 1000, 2000, and 3000 rpm with the regenerator: (1) against the inner shoulder on top of the cooler, (2) against the face in which the heater tubes are brazed, (3) against the top face of the cooler, and (4) midway between the cooler and the heater tube entrances as shown in Figure 9. At the conclusion of the foregoing part of the test, the entrances and exits of the heater tubes and the tubes in the cooler were rounded off and the pressure differentials were again obtained with the regenerator against the face containing the heater tube entrances.

The pressure differentials for all of the conditions checked are shown in Table I. This data indicates that the pressure differential, at any particular location or speed, did not change appreciably regardless of the regenerator location or the condition of the tube entrance and discharge.

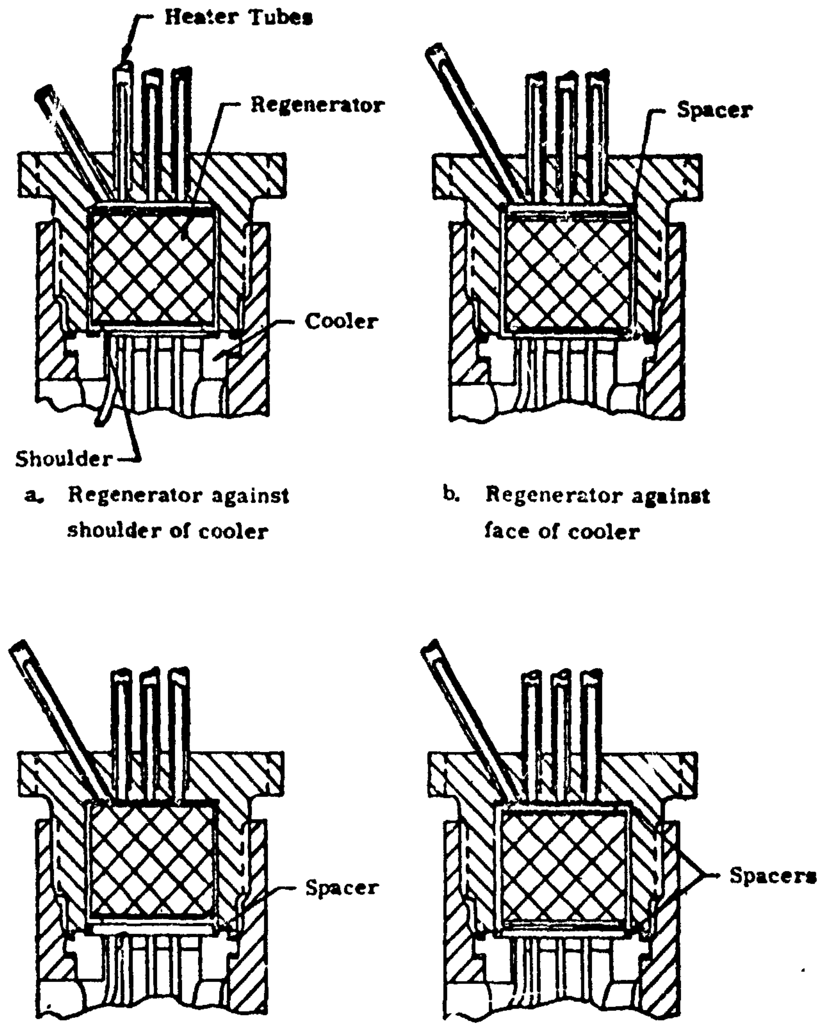
A phenomena worth noting is the decrease in Δ P at any location with an increase in piston speed instead of an increase in Δ P.

As was pointed out in the last monthly progress report, this condition is caused by restrictions in the connections to the pressure transducers. An attempt was made to rectify this problem, but due to the physical shape of the test apparatus the transducer inlet tube diameters could not be increased and the dead space decreased significantly to eliminate it entirely. Consequently, the results obtained were purely qualitative; but, as there were not significant changes noted, it was not necessary to pursue the restriction problem further.

In order to verify the fact that there were no appreciable changes in pressure differentials, Bourdon gages were connected to the pressure taps and steady flow readings were obtained for a range of air flow from 0 26 lb/min to 0.57 lb/min.

The results obtained for the regenerator against the cooler face, for the regenerator midway between the cooler and the face in which the heater tubes are brazed, and for the regenerator against the heater tube entrance face are shown in Table II. Although the pressure differentials are slightly higher when the regenerator is displaced from the middle position, the penalty for doing so is not large enough to be of concern.

It may be concluded for the Unit Heat Exchanger Test that the position of the regenerator does not change the pressure differentials appreciably regardless of whether the flow is fluctuating or constant and that the effect of entrance and exit losses due to sharp corner conditions at tube openings was not apparent.



a. Regenerator against shoulder of cooler

b. Regenerator against face of cooler

c. Regenerator against heater tube entrance face

d. Regenerator midway between cooler and heater tube entrance

Figure 9. Locations of Regenerator for Unit Heat Exchanger Test

TABLE I.

Pressure Differentials in Unit Heat Exchanger for Fluctuating Flow

Location of Regenerator	CPM	ΔP at Top of Cylinder (psi)	ΔP at Bottom of Cooler (psi)	ΔP Between Cooler and Regenerator (psi)	ΔP at Heater Tube Entrance (psi)
Regenerator against shoulder of cooler	1000	2.2	2.4	2.0	0.2
	2000	1.3	2.2	1.8	0.2
	3000	0.7	1.4	1.0	0.2
Regenerator against heater tube entrance face	1000	2.2	2.3	2.0	0.2
	2000	1.2	0.2	1.8	0.2
	3000	0.7	1.4	1.1	0.2
Regenerator against top face of cooler	1000	2.3	2.3	2.0	0.2
	2000	1.3	2.0	1.6	0.2
	3000	0.6	1.3	1.0	0.2
Regenerator midway between cooler and heater tube entrance face	1000	2.3	2.3	2.0	0.2
	2000	1.5	2.0	1.8	0.2
	3000	0.8	1.4	1.0	0.2
Regenerator against heater tube entrance face with rounder tube entrances and exits	1000	2.2	2.4	2.0	0.2
	2000	1.6	2.0	1.7	0.2
	3000	0.7	1.4	1.0	0.2

TABLE II.

Pressure Differentials Across Heat Exchanger Components for Steady Flow

Location of Regenerator	*Orifice ΔP (In. Kerosene)	W (lb/min)	Inlet Pressure (psi)	Total** Sidearm ΔP (In. Hg)	ΔP Across Regenerator (In. Hg)	ΔP Across Heater (In. Hg)	ΔP Across Cooler (In. Hg)
Regenerator against face of cooler	80	0.578	34.5	65.0	19.9	9.80	32.25
	70	0.516	30.0	56.4	17.95	7.95	28.5
	60	0.457	26.0	48.7	16.10	6.40	24.14
	50	0.397	22.0	40.8	13.85	4.95	20.70
	40	0.331	17.0	32.0	11.60	3.40	16.07
	30	0.266	13.0	24.5	9.20	2.40	11.83
Regenerator midway between cooler and heater tube entrance face	80	0.564	32.0	60.4	21.0	9.40	27.65
	70	0.504	28.0	53.1	19.0	7.80	24.93
	60	0.444	24.0	45.0	16.7	6.10	19.98
	50	0.3851	20.0	37.8	14.4	4.70	17.05
	40	0.3251	16.0	30.0	12.35	3.40	12.75
	30	0.2641	12.5	23.0	9.8	2.40	11.23
Regenerator against heater tube entrance face	80	0.564	32.0	61.0	19.6	11.40	27.57
	70	0.504	28.0	53.0	17.8	9.40	23.10
	60	0.444	24.0	45.8	16.0	7.60	20.40
	50	0.385	20.0	37.8	13.75	5.75	16.34
	40	0.326	16.5	30.2	11.7	4.30	13.03
	30	0.264	12.5	23	9.5	2.90	10.00

*Orifice Size = 0.200 for all three conditions

**Includes pressure drop across connecting duct between cylinder and bottom of cooler

III. MODEL PD-67 STIRLING ENGINE DESIGN

During the month of September, the design of the Model PD-67 Stirling engine was continued. The design layouts are currently in the final stage of completion and any changes are limited to those where refinements can be achieved. The detailed work performed during this reporting period is described in the following paragraphs.

CYLINDER HEAD DESIGN

The Model PD-67 engine cylinder head design has been changed from that shown in previous reports. The modifications consisted of adapting a new heater section to the main cylinder housing. See Figure 10. This revision was found necessary, based on test results obtained from the running of the Model PD-46 engine, in order that the heat transfer characteristics could be improved. With the new cylinder head configuration, it is anticipated that an increase in engine output and efficiency can be achieved.

The new design incorporates 76 heater tubes arranged in two circumferential rows. The previous PD-67 design had 96 tubes arranged in three rows. The tube ID has also been reduced from 0.072 to 0.060 inches and the length increased from 4.1 to 6 inches. By this change higher Reynolds number and greater tube l/d are achieved which should improve the heat transfer coefficient. The new design more nearly matches the tube proportions used in similar Philip's Stirling engine designs. The NaK manifold has also been modified to accept the new heater tube configuration. An inlet manifold, identical to the outlet manifold, has been incorporated. This revision simplifies the baffling arrangement and the center thermocouple tube connection. NaK flow distribution is maintained by utilizing 12 small holes located in the partition plate. The average NaK velocity in the tubes region is approximately three ft/sec and the pressure drop is estimated to be three psi.

DRIVE MECHANISM DESIGN

The drive mechanism design has not changed from that shown in previous reports. The connecting rod and main bearings have been finalized. The calculated B-10 operating life is approximately 100,000 hours. The crankshaft curvic coupling has been solidified at 24 teeth. Currently the design is being checked for critical stress and balance. At completion of this work, the stack layout will be finalized and this major engine subassembly will be ready for detailing and checking.

CRANKCASE DESIGN

The crankcase design layout is currently in process. During this reporting period the zero gravity oil cooler tube manifold assembly has been designed and the case inner wall configuration established. At the present time, the oil pick-up tubes and the bearing mounting plates are being developed.

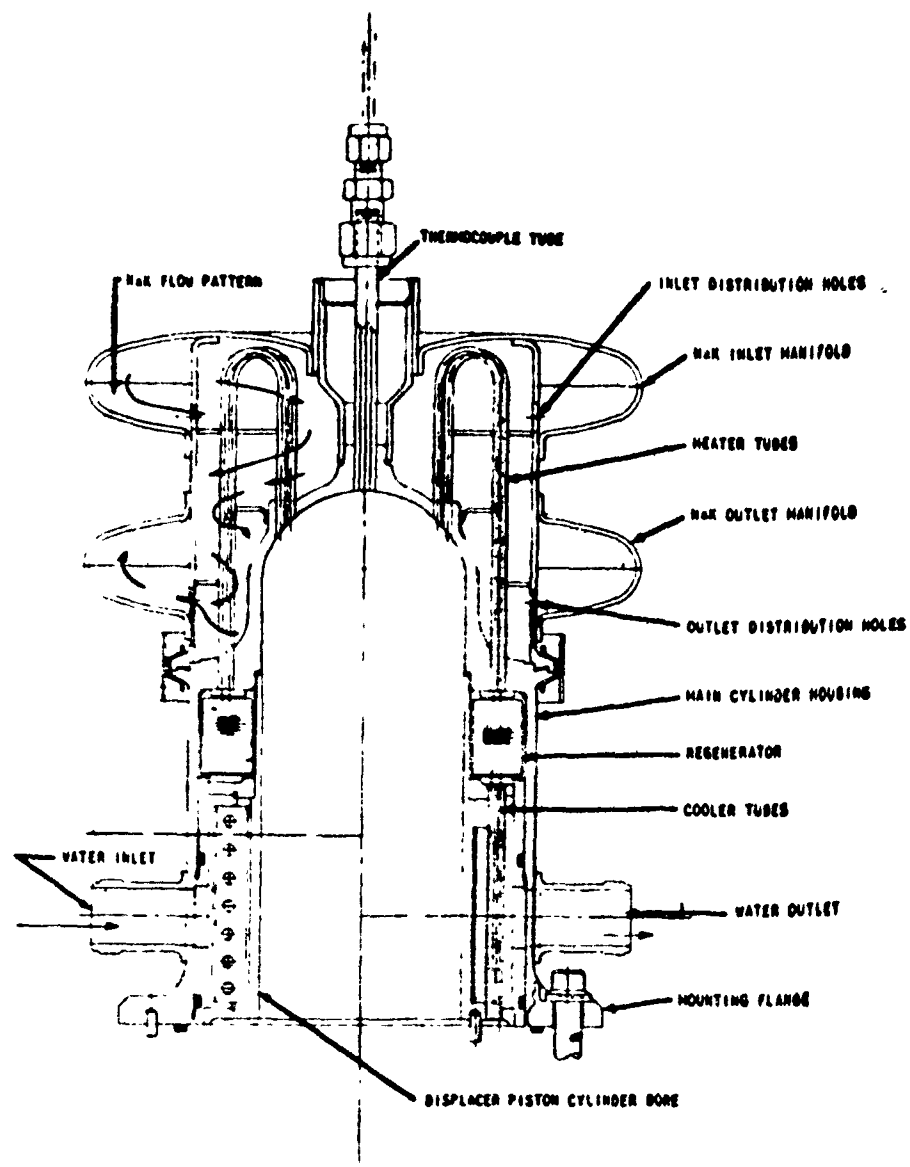


Figure 10. Modified PD-67 Cylinder Head Design

ALLISON

A heat transfer investigation is in progress to determine if the cooler manifold assembly is adequate to maintain a desired oil temperature level. This work will be completed shortly.